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GB 2135256 A

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Online databases: WPI, CLAIMS.

(54) **Torque distribution control using continuously variable transmission in parallel with differential**

(57) A torque distribution control apparatus for use on a motor vehicle controls torques to be distributed to drive road whee of the motor vehicle depending on running conditions of the motor vehicle. The torque distribution control apparatus (11) includes a differential (1) having first and second output shafts (6, 7) and means for distributing power from a drive source the motor vehicle to the first and second output shafts, a continuously variable transmission (20) having first and second transmission shafts (24, 26) and disposed parallel to the differential, the continuously variable transmission including mear for transmitting rotation of the first transmission shaft to the second transmission shaft or transmitting rotation of the secon transmission shaft to the first transmission shaft at a continuously variable rotational speed, first coupling means (12) for coupling the first output shaft (6) and the first transmission shaft (24) to each other, second coupling means (13) for coupli the second output shaft (7) and the second transmission shaft (26) to each other, and transmission ratio control means for controlling the transmission ratio of the continuously variable transmission depending on running conditions of the motor vehicle.

The continuously variable transmission is preferably a hydrostatic swash plate pump/motor arrangement controlled in dependence on steering angle, vehicle speed and lateral acceleration.

FIG.1

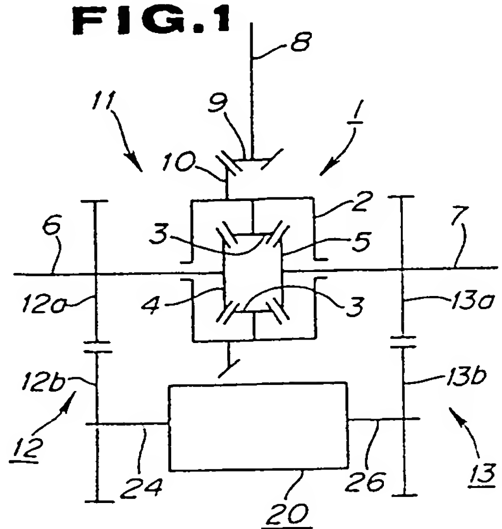


FIG.1

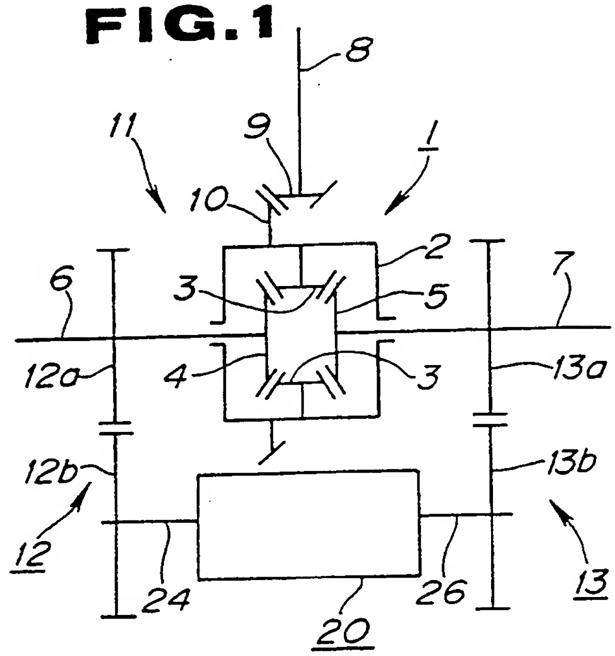


FIG.4

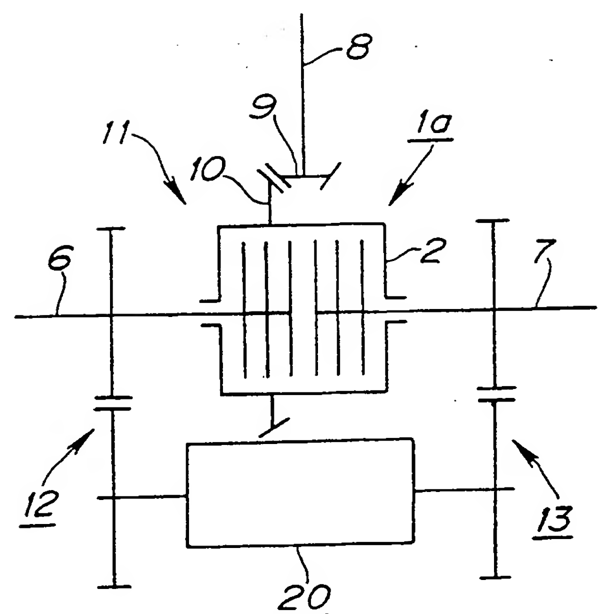


FIG.5

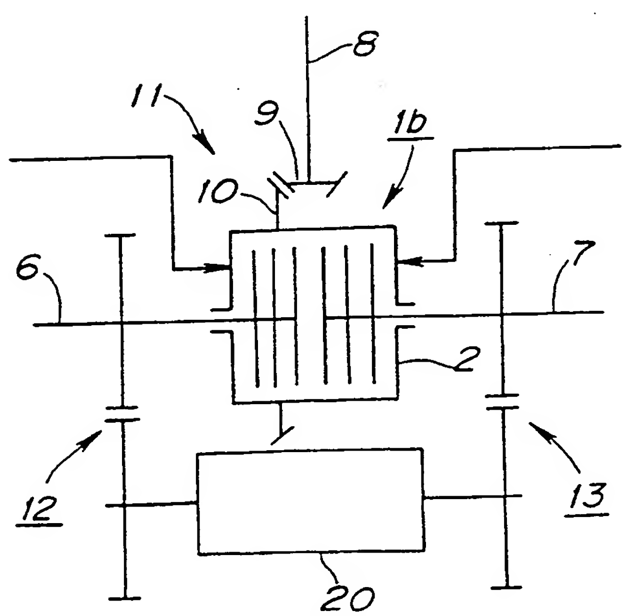


FIG.2

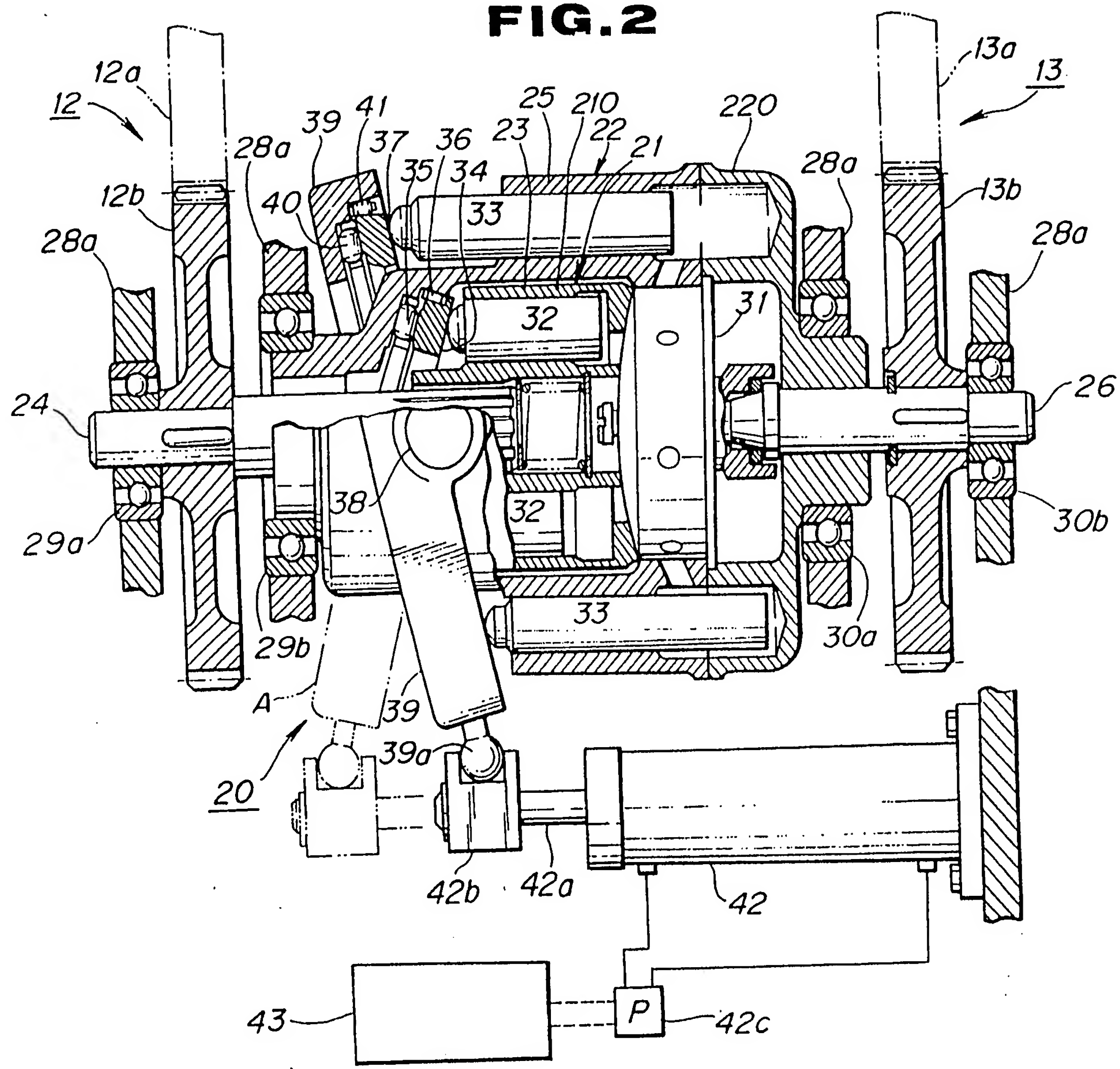

12
12a
12b
28a
39
41
37
36
35
34
23
25
210
22
21
220
28a
13a
13
13b
28a
33
32
31
24
26
38
32
29a
30b
29b
33
30a
A
39
39a
20
42b
42a
42
43
P
42c

FIG. 3a

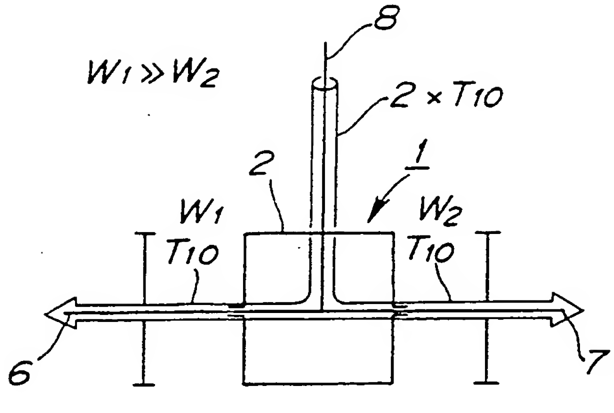


FIG. 3b

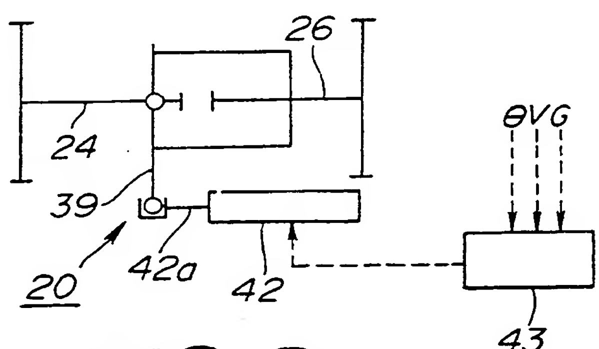


FIG. 3c

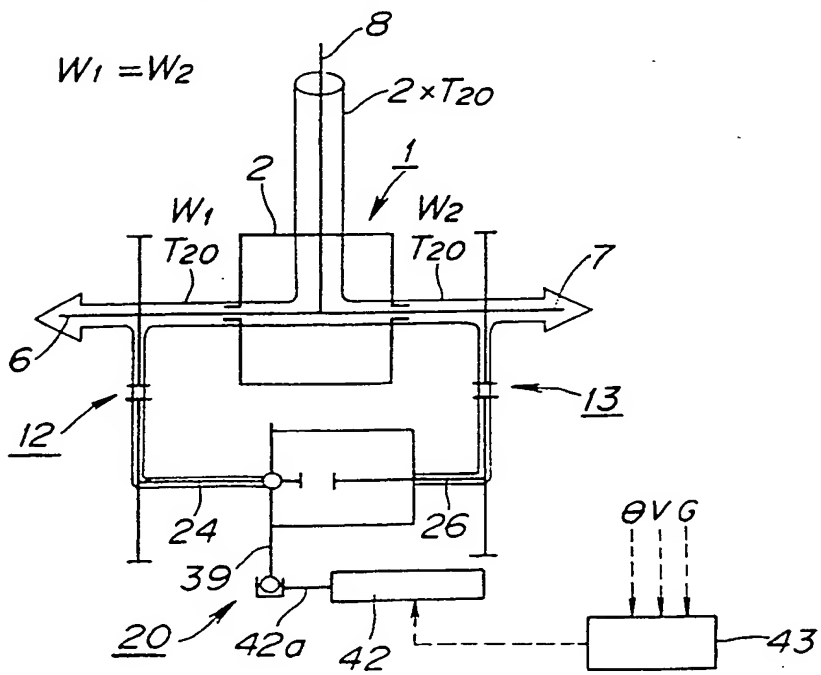


FIG. 3d

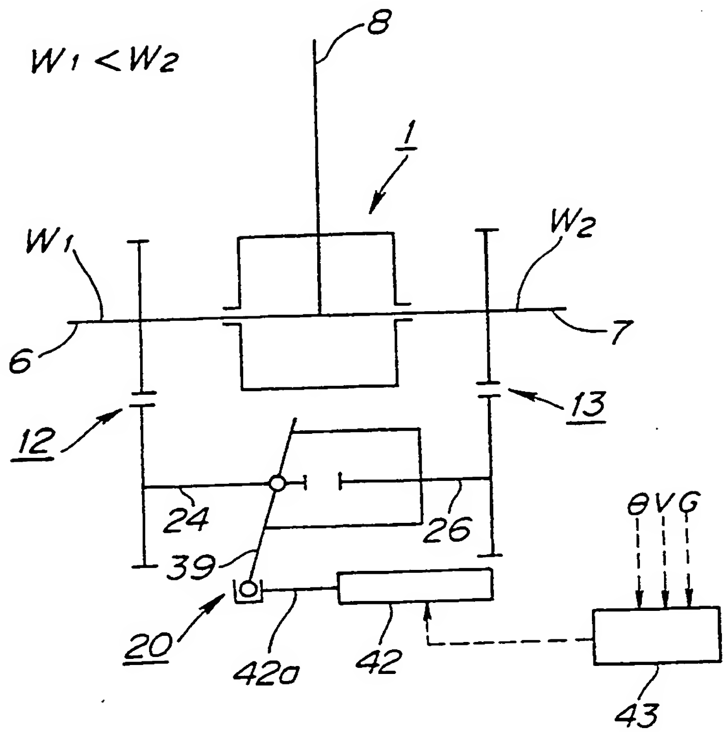
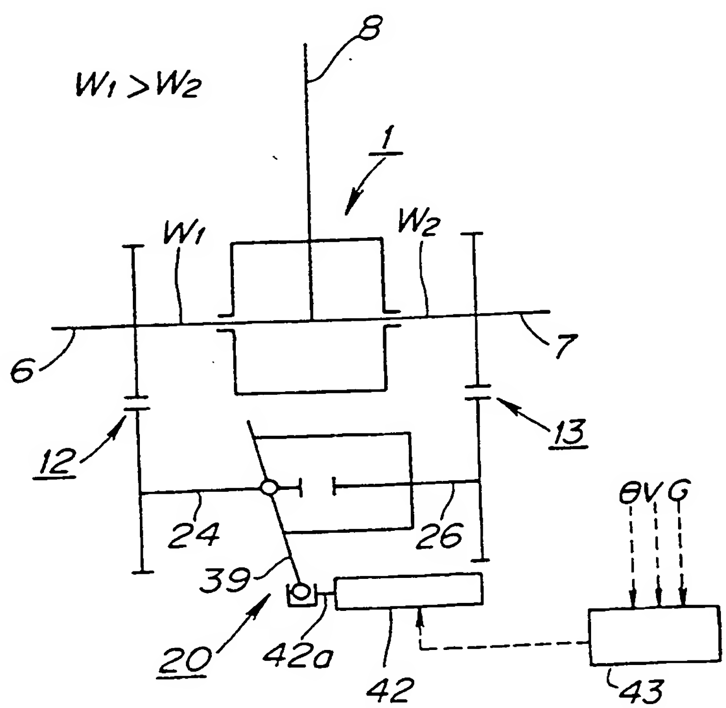

W1 < W2
8
1
W1
W2
6
7
12
13
24
26
39
θ V G
20
42a
42
43

FIG. 3e


W1 > W2
8
1
W1
W2
6
7
12
13
24
26
39
θ V G
20
42a
42
43

TORQUE DISTRIBUTION CONTROL APPARATUS FOR MOTOR VEHICLE

The present invention relates to an apparatus for controlling torques distributed to drive wheels of a motor vehicle depending on running conditions of the motor vehicle.

It is well known to provide the power transmission mechanism of a motor vehicle with a differential for allowing left and right or front and rear road wheels to rotate at different speeds thereby to permit the motor vehicle to make a turn smoothly.

For example, a motor vehicle has a differential disposed between left and right drive road wheels. Such a differential is designed to transmit equal torques to the left and right drive road wheels at all times. Therefore, when one of the drive road wheels slips on a road surface having a lower coefficient of friction, such as a muddy, sandy, or frozen road surface, the torque transmitted to the other drive road wheel is greatly reduced. At this time, the drive power cannot effectively be transmitted to the road surface, and the motor vehicle fails to move from the slippery road surface.

In order to solve the aforesaid problem of the conventional differential, there have been proposed a limited-slip differential and a torque distribution control apparatus as disclosed in Japanese Laid-Open Patent Publication No. 62-203826.

The torque distribution control apparatus disclosed in the above publication comprises a torque distributor disposed in bypassing relation to a central differential on a four-wheel-drive motor vehicle. The torque distributor includes a first gearing coupled to one side gear of a central differential, second and third gearings coupled to the other side gear of the central differential and having gear ratios larger than the gear ratio of the first gearing, a

selector clutch operatively coupled to the driven gears of the second and third gearings for selecting one of the second and third gearings at a time, and a variable-torque hydraulic clutch for connecting the driven gear of the first gearing and the selector clutch to each other.

With this arrangement, the torque applied to one or the other of front and rear road wheels is increased, and the ratio of the torque applied to one road wheel to the torque applied to the other road wheel is continuously varied to improve maneuverability and stability of the motor vehicle.

The disclosed torque distribution control apparatus is quite complex in structure as it comprises the first, second, and third gearings, the selector clutch, and the hydraulic clutch. The hydraulic clutch comprises a number of clutch discs and hence is large in diameter. Therefore, the torque distribution control apparatus is large in size.

According to the present invention, there is provided a torque distribution control apparatus for use on a motor vehicle having a drive source, comprising a differential having first and second output shafts and means for distributing power from the drive source to the first and second output shafts, a continuously variable transmission having first and second transmission shafts and disposed parallel to the differential, the continuously variable transmission including means for transmitting rotation of the first transmission shaft to the second transmission shaft or transmitting rotation of the second transmission shaft to the first transmission shaft at a continuously variable rotational speed, first coupling means for coupling

the first output shaft and the first transmission shaft to each other, second coupling means for coupling the second output shaft and the second transmission shaft to each other, and transmission ratio control means for controlling the transmission ratio of the continuously variable transmission depending on running conditions of the motor vehicle.

Some embodiments of the invention will now be described by way of example and with reference to the accompanying drawings, in which:-

FIG. 1 is a schematic diagram of a torque distribution control apparatus according to an embodiment of the present invention;

FIG. 2 is a cross-sectional view of a hydraulically operated continuously variable transmission;

FIGS. 3a through 3e are schematic diagrams showing the manner in which the torque distribution control apparatus shown in FIG. 1 operates;

FIG. 4 is a schematic diagram of a torque distribution control apparatus according to another embodiment of the present invention; and

FIG. 5 is a schematic diagram of a torque distribution control apparatus according to still another embodiment of the present invention.

A torque distribution control apparatus according to an embodiment of the present invention will be described below with reference to FIGS. 1, 2, and 3a through 3e.

As shown in FIG. 1, the torque distribution control apparatus, generally indicated by the reference numeral 11, for use on a motor vehicle comprises a differential 1, a hydraulically operated continuously variable transmission (CVT) 20 disposed parallel to the differential 1, a first coupling means 12 coupling a first output shaft 6 and a

first transmission shaft 24 of the CVT 20, and a second coupling means 13 coupling a second output shaft 7 and a second transmission shaft 26 of the CVT 20.

The differential gear 1 comprises a ring gear 10 held in mesh with a gear 9 mounted on one end of a power transmission shaft 8, a differential case 2 fixed to and rotatable with the ring gear 10, a pair of pinions 3 rotatably mounted in the differential case 2, and a pair of side gears 4, 5 disposed in the differential case 2 and meshing with the pinions 3, respectively. The first output shaft 6 is connected to the side gear 4, and the second output shaft 7 is connected to the side gear 5.

Output power from an engine (not shown) is varied in its rotational speed by a transmission (not shown), and applied through the power transmission shaft 8 and the gear 9 to the ring gear 10 of the differential 1. The power applied to the ring gear 10 is then transmitted through the pinions 3 to the side gears 4, 5, from which it is distributed to the first and second output shafts 6, 7.

The first coupling means 12 comprises a pair of intermeshing spur gears 12a, 12b, and the second coupling means 13 comprises a pair of intermeshing spur gears 13a, 13b.

As shown in FIG. 2, the CVT 20 has two hydraulic pump/motors 22, 21, one operating as a pump and the other as a motor. For the purpose of illustration only, the outer hydraulic pump/motor 22 will hereinafter be described as a pump and the inner hydraulic pump/motor 21 as a motor.

The hydraulic pump 22 comprises a cylinder unit 220 having a plurality of pump cylinders 25, a plurality of pump plungers 33 slidably inserted respectively in the pump cylinders 25, a pump swash plate 37 held against the tip ends of the pump plungers 33, and a swash plate support 39 supporting the pump swash plate 37. The cylinder unit 220 is rotatably supported in a casing 28a by means of bearings

29a, 29b, 30a, 30b, and has a rear end connected to the second transmission shaft 26 for rotation therewith. The pump cylinders 25 extend parallel to each other and are disposed concentrically along the outer circumference of the cylinder unit 220 at circumferentially equally spaced intervals. The cylinder unit 220 has a hollow space therein with the hydraulic pump 21 housed therein.

The pump plungers 33 inserted respectively in the pump cylinders 25 are reciprocally movable in the axial direction thereof, and have their tip ends held against the pump swash plate 37. The pump swash plate 37 is rotatably supported on the swash plate support 39 by means of a thrust bearing 40 and a radial bearing 41. The swash plate support 39 is pivotally supported on the casing 28a or a structural member fixed to a vehicle body by means of a pair of pivot shafts 38 on the opposite sides of the swash plate support 39. Therefore, the angle at which the pump swash plate 37 is inclined can be varied by tilting the swash plate support 39.

A hydraulic cylinder 42 is mounted on the casing 28a and has a piston rod 42a operatively coupled to the swash plate support 39. More specifically, a ball 39a on an end of the swash plate support 39 is fitted in a forked member 42b fixed to the distal end of the piston rod 42a. The swash plate support 39 is thus angularly movable about the pivot shafts 38 in response to extension or contraction of the hydraulic cylinder 42. The hydraulic cylinder 42 is operated by a hydraulic unit 42c which is controlled by a controller 43.

The hydraulic motor 21 comprises a cylinder unit 210 having a plurality of motor cylinders 23, a plurality of motor plungers 32 slidably inserted respectively in the motor cylinders 23, and a motor swash plate 34 held against the tip ends of the motor plungers 23. The cylinder unit 210 is splined to the first transmission shaft 24 for rota-

tion therewith. The motor cylinders 23 extend parallel to each other and are disposed concentrically along the outer circumference of the cylinder unit 210 at circumferentially equally spaced intervals. The cylinder unit 210 has a rear surface held relatively rotatable against the front surface of a distribution disc 31. The distribution disc 31 is fixed to the cylinder unit 220, and has oil passages through which the cylinders of the hydraulic pump 22 and the cylinders of the hydraulic motor 21 communicate with each other.

The motor plungers 32 inserted respectively in the motor cylinders 23 are reciprocally movable in the axial direction thereof, and have their tip ends held against the motor swash plate 34 which is inclined at an angle to the axis of the cylinder unit 210. The motor swash plate 34 is rotatably supported on an inner wall of the cylinder unit 220 by means of a thrust bearing 35 and a radial bearing 36.

It is assumed that the piston of the hydraulic cylinder 42 is retracted to hold the swash plate support 39 and the swash plate 37 at a maximum angular position indicated by the solid lines in FIG. 2. When the second transmission shaft 26 is rotated by the second coupling means 13, the cylinder unit 220 is rotated with respect to the pump swash plate 37, and the pump plungers 33 are reciprocally moved axially while being held in contact with the pump swash plate 37. Working oil discharged from those of the pump cylinders 25 which house the pump plungers 33 in the discharge stroke flows through the distribution disc 31 into those of the motor cylinders 23 which house the motor plungers 32 in the expansion stroke, applying a forward thrust to these motor plungers 32. Working oil discharged from those motor cylinders 23 which house the motor plungers 32 in the discharge stroke flows through the distribution disc 31 back into the pump cylinders 25 which house the pump plungers 33 in the suction stroke.

The thrust applied to the motor plungers 32 is divided by the motor swash plate 34 into a force normal to

the surface thereof and a force parallel thereto. The latter force then imposes a torque to the cylinder unit 210 relatively to the cylinder unit 220, and the torque is transmitted from the cylinder unit 210 to the first transmission shaft 24.

When the pump swash plate 37 is inclined to its maximum angular position, as described above, the stroke by which the pump plungers 33 are axially moved is maximum, and hence the amount of working oil discharged from the hydraulic pump 22 is also maximum. If it is assumed at this time that the rotational speed of the second transmission shaft (input shaft) 26 is indicated by a and the rotational speed of the first transmission shaft (output shaft) 24 is indicated by b, then the ratio α of the rotational speed a to the rotational speed b, i.e., the transmission ratio α, is expressed by $\alpha = a : b$ and is smaller than 1 since $b > a$. The torque applied to the first transmission shaft 24 is now minimum.

When the pump swash plate 37 is not inclined, i.e., lies perpendicularly to the pump plungers 33, the stroke of any sliding movement of the pump plungers 33 is zero, and hence no working oil is discharged from the hydraulic pump 22. The second and first transmission shafts 26, 24 rotate at the same speed. Thus, the transmission ratio α is $\alpha = 1 : 1$.

The pump swash plate 37 is then inclined by the hydraulic cylinder 42 to its opposite symmetrical maximum angular position as indicated by the broken lines A. If it is assumed at this time that the rotational speed of the second transmission shaft 26 is indicated by b and the rotational speed of the first transmission shaft 24 is indicated by a, then the ratio α of the rotational speed b to the rotational speed a, i.e., the transmission ratio α, is expressed by $\alpha = b : a$ and is larger than 1 since $b > a$. The torque applied to the first transmission shaft 24 is now maximum.

As described above, the CVT 20 transmits the rotation of the second transmission shaft 26 to the first transmission shaft 24 at a reduced, equal, or increased speed depending on the angle at which the pump swash plate 37 is inclined. Conversely, the CVT 20 can transmit the rotation of the first transmission shaft 24 to the second transmission shaft 26 at a reduced, equal, or increased speed. At this time, the hydraulic pump/motor 21 operates as a pump and the hydraulic pump/motor 22 as a motor. Accordingly, the CVT 20 operates as a transmission for transmitting rotative power at a reduced, equal, or increased speed from one of the first and second transmission shafts 24, 26 as an input shaft to the other shaft as an output shaft.

The controller 43 receives input signals representing a steering angle θ, a vehicle speed V, and a lateral acceleration G applied to the vehicle body, and establishes optimum rotational speeds W_1, W_2 for the first and second output shafts 6, 7 depending on the running condition of the motor vehicle. The controller 43 then actuates the hydraulic cylinder 42 to tilt the swash plate 39 to control the transmission ratio of the CVT 20.

The hydraulically operated CVT 20 is a relatively simple construction in which the hydraulic pump 22 and the hydraulic motor 21 are connected to each other by hydraulic passages. The first and second transmission shafts 24, 25 are disposed coaxially to each other, and the hydraulic pump/motors 21, 22 are also positioned coaxially to each other. Therefore, CVT 20 is simple in structure, small in size, has rotatable members of small inertia, and can be installed with greater freedom.

Operation of the torque distribution control apparatus 11 will be described below.

As shown in FIG. 1, the power from the engine is applied through the power transmission shaft 8 and the gear

9 to the ring gear 10 of the differential 1, by which the power is distributed to the first and second output shafts 6, 7 and hence to road wheels (not shown) coupled thereto.

It is assumed here that the torque distribution control apparatus 11 is not present. The torque T_0 and the rotational speed W_0 of the ring gear 10, the torque T_1 and the rotational speed W_1 of the first output shaft 6, and the torque T_2 and the rotational speed W_2 of the second output shaft 7 have the following relationship:

$$T_1 = T_2 = 1/2\ T_0$$

$$1/2\ (W_1 + W_2) = W_0$$

If the road wheel coupled to the first output shaft 6 is stuck in a muddy road and the frictional resistance applied from the road to the road wheel is greatly reduced, the first output shaft 6 is subjected to only the frictional resistance from the road and a frictional force from a bearing (not shown), and the torque of the road wheel is also greatly reduced. The frictional resistance applied from the road surface to the road wheel coupled to the second output shaft 7 is sufficiently large.

Therefore, since the second output shaft 7 is subjected to a greater resistance and the first output shaft 6 to a smaller resistance, the pinions 3 meshing with the side gear 4 roll on the pinions 3, allowing the first output shaft 6 to rotate at a higher speed and reducing the torque of the first output shaft 6. Inasmuch as the same torque is applied to the first and second output shafts 6, 7, the torque applied to the second output shaft 7 is also reduced. Consequently, the engine power is consumed to rotate the road wheel coupled to the first output shaft 6, and is not effectively transmitted to the road surface. This condition will be described in greater detail with reference to FIG. 3a. The rotational speed W_1 of the first output shaft 6 is much larger than the rotational speed W_2 of the second output shaft 7. The load torque T_{10} acting on the first output

shaft 6 is reduced, and the same load torque T_{10} is applied to the second output shaft 7. Therefore, the torque 2 × T_{10} of the ring gear 10 is of a small value.

The torque distribution control apparatus 11 operates as shown in FIGS. 3b through 3d.

When the motor vehicle runs straight ahead, the hydraulic cylinder 42 of the torque distribution control apparatus 11 is operated to eliminate the angle of inclination of the pump swash plate 37, i.e., to keep the pump swash plate 37 perpendicular to the pump plungers 33, so that the transmission ratio of the CVT 20 is 1 : 1 for transmitting the rotative power at the same speed to the first and second transmission shafts 24, 26. If there is developed a difference in speed between the first and second transmission shafts 24, 26, e.g., if the second output shaft 7 rotates at a higher speed than the first output shaft 6, then the rotation of the second output shaft 7 is applied through the second coupling means 13 to the CVT 20 to drive the hydraulic motor 21 for thereby producing a large load torque.

The large load torque T_{20} thus produced is then transmitted through the CVT 20, the first transmission shaft 24, and the first coupling means 12 to the first output shaft 6, thus increasing the torque of the first output shaft 6.

As a result, the rotational speed W_1 of the first output shaft 6 and the rotational speed W_2 of the second output shaft 7 become substantially the same as each other, and the torque of the first and second output shafts 6, 7 becomes the large torque T_{20}. The casing 2 integral with the ring gear 10 has a torque 2 × T_{20}, and the output power from the engine can effectively be utilized.

When the motor vehicle makes a left turn (the upper side shown in FIGS. 3a through 3e is the front side of the motor vehicle), the controller 43 controls the transmission

ratio of the CVT 20 so that the relationship between the rotational speeds W_1, W_2 ($W_2 > W_1$) determined by the radius of the cornering circle will be maintained.

More specifically, the controller 43 actuates the hydraulic cylinder 42 to move the piston rod 42a for angularly moving the swash plate support 39 of the CVT 20, so that the ratio of the rotational speed W_2 of the second transmission shaft 26 to the rotational speed W_1 of the first transmission shaft 24, i.e., the transmission ratio of the CVT 20, will be $W_2 : W_1$. The motor vehicle is thus allowed to turn smoothly.

If the lefthand road wheel slips during the left turn, the torque distribution control apparatus 11 applies a braking torque to the output shaft coupled to the slipping road wheel, thus keeping the desired rotational speeds W_1, W_2.

FIG. 3e shows operation of the torque distribution control apparatus 11 while the motor vehicle is making a right turn. The torque distribution control apparatus 11 controls the rotational speeds W_1, W_2 such that $W_1 > W_2$ in a manner similar to the operation described with reference to FIG. 3d.

As described above, the torque distribution control apparatus 11 can transmit engine power to the road wheels with a desired torque distribution regardless of whether the motor vehicle runs straight ahead, makes a left turn, or a right turn.

FIG. 4 shows a torque distribution control apparatus 11 according to another embodiment of the present invention, the torque distribution control apparatus 11 comprising a viscous coupling differential 1a, a CVT 20, and coupling means 12, 13.

FIG. 5 illustrates a torque distribution control apparatus 11 according to still another embodiment of the present invention, the torque distribution control apparatus 11 comprising a hydraulically operated clutch differential 1b, a CVT 20, and coupling means 12, 13.

Each of the first and second coupling means 12, 13 in the above embodiments is not limited to a gear train, but may comprise a chain and sprocket mechanism, a belt and pulley mechanism, or the like.

The CVT 20 is not limited to a hydraulically operated CVT, or may be a friction wheel type CVT, a V-belt type CVT, a chain type CVT, a traction drive type CVT, or the like.

With the present invention, as described above, the distribution of torques to drive wheels can smoothly be controlled by a simple construction, and the motor vehicle incorporating the torque distribution control apparatus of the invention is made compact and has better maneuverability.

Although there have been described what are at present considered to be the preferred embodiments of the present invention, it will be understood that the invention may be embodied in other specific forms without departing from the essential characteristics thereof. The present embodiments are therefore to be considered in all aspects as illustrative, and not restrictive. The scope of the invention is indicated by the appended claims rather than by the foregoing description.

It will thus be seen that the present invention, at least in its preferred forms, provides a torque distribution control apparatus for motor vehicles, which is relatively simple in structure, small in size, and can smoothly control the distribution of torques to drive road wheels.

It is to be clearly understood that there are no particular features of the foregoing specification, or of any claims appended hereto, which are at present regarded as being essential to the performance of the present invention, and that any one or more of such features or combinations thereof may therefore be included in, added to, omitted from or deleted from any of such claims if and when

amended during the prosecution of this application or in the filing or prosecution of any divisional application based thereon. Furthermore the manner in which any of such features of the specification or claims are described or defined may be amended, broadened or otherwise modified in any manner which falls within the knowledge of a person skilled in the relevant art, for example so as to encompass, either implicitly or explicitly, equivalents or generalisations thereof.

CLAIMS:

1. A torque distribution control apparatus for use on a motor vehicle having a drive source, comprising:

a differential having first and second output shafts and means for distributing power from the drive source to said first and second output shafts;

a continuously variable transmission having first and second transmission shafts and disposed parallel to said differential, said continuously variable transmission including means for transmitting rotation of said first transmission shaft to said second transmission shaft or transmitting rotation of said second transmission shaft to said first transmission shaft at a continuously variable rotational speed;

first coupling means for coupling said first output shaft and said first transmission shaft to each other;

second coupling means for coupling said second output shaft and said second transmission shaft to each other; and

transmission ratio control means for controlling the transmission ratio of said continuously variable transmission depending on running conditions of the motor vehicle.

2. A torque distribution control apparatus according to claim 1, wherein said continuously variable transmission comprises a hydraulically operated transmission having a pair of first and second oil pump/motors.

3. A torque distribution control apparatus according to claim 2, wherein said first oil pump/motor is connected to said first transmission shaft and said second oil pump/motor is connected to said second transmission shaft, said second oil pump/motor being disposed coaxially around said first oil pump/motor.

4. A torque distribution control apparatus according to claim 3, wherein each of said first and second oil pump/motors comprises a cylinder unit having a plurality of cylinders, a plurality of plungers axially slidably disposed in said cylinders, respectively, and a swash plate held against tip ends of said plungers, said swash plate of said second oil pump/motor being tiltably mounted on a vehicle body of the motor vehicle, said swash plate of said first oil pump/motor being rotatably mounted in said cylinder unit of the second oil pump/motor.

5. A torque distribution control apparatus according to any preceding claim, wherein said differential comprises a gear differential.

6. A torque distribution control apparatus according to any of claims 1 to 4, wherein said differential comprises a viscous coupling differential.

7. A torque distribution control apparatus according to any of claims 1 to 4, wherein said differential comprises a hydraulically operated clutch differential.

8. A torque distribution control apparatus according to any preceding claim, wherein said transmission ratio control means comprises means for controlling the transmission ratio of said continuously variable transmission depending on the speed of travel and the steering angle of the motor vehicle.

9. A torque distribution control apparatus according to any of claims 1 to 7, wherein said transmission ratio control means comprises means for controlling the transmission ratio of said continuously variable transmission depending on the speed of travel of the motor vehicle and the lateral acceleration applied to the motor vehicle.

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